Pelican Hydroelectric Project
Hydraulic Transient Analysis

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1 Introduction

The Pelican hydroelectric plant was built in the early 1940's and provides a majority of the electrical power for the City of Pelican, Alaska.[1] The original 400 kW plant burned in 1982 and was rebuilt and upgraded to a 600 kW synchronous generator and 100 kW induction generator.[2] This hydraulic transient study was undertaken in support of the penstock and flume replacement being designed by Polarconsult, Alaska, Inc.[3] Here we calculate the minimum closing times to limit the hydraulic transients to 20% over the maximum static head for a new 36 in steel penstock and for design options removing the surge tank and replacing the open flume with pressurized steel pipe. We also calculate how removing the surge tank will affect the plant's speed governing capability. The nomenclature used in this report is given in Table 1.

2 Summary of Results

With the surge tank retained, the minimum closing time for a linear decrease in total plant discharge coefficient from a total discharge of $80\,\mathrm{ft}^3/\mathrm{s}$ to zero

A	cross sectional area of pipe (m^2)
a	pressure wave propagation velocity ($\mathrm{m/s}$)
D	outside diameter of pipe (m)
E	modulus of elasticity ($\mathrm{N/m}^2$ or Pa)
e	pipe wall thickness (m)
g	gravitational acceleration ($\mathrm{m/s}^2$)
H	head (m)
I	moment of inertia ($\mathrm{kg\,m}^2$)
K_E	effective bulk modulus of pipe ($\mathrm{N/m}^2$ or Pa)
K_V	volume modulus of pipe ($\mathrm{N/m}^2$ or Pa)
K_W	bulk modulus of water ($\mathrm{N/m}^2$ or Pa)
L	pipe length (m)
N	rotational speed (rpm)
P	power (W)
Q	flow (m^3/s)
T_2	time for pressure wave to travel twice length of pipe (s)
T_G	gate closing time (s)
T_M	machine starting time (s)
T_W	water starting time (s)
V	water velocity ($\mathrm{m/s}$)
ρ	density of water ($\mathrm{kg/m}^3$)
σ	pipe-line constant (dimensionless)
ω	rotational speed ($\mathrm{rad/s}$)

Table 1: Nomenclature for hydraulic transient analysis.

is about 4.8 s. With the surge tank removed and the open flume replaced with 36 in steel pipe, the minimum closing time is about 21 s. With the open flume replaced with 42 in steel pipe, the minimum closing time is about 20 s. The present closing time of about 10 s probably results in speeds close to the maximum turbine runaway speed. The longer closing time for the pressurized pipeline would extend the overspeed time.

With the surge tank retained, the plant will be capable of picking up a 136 kW load step with a 6 Hz frequency deviation. With the open flume replaced with 42 in steel pipe and no surge tank, the plant will be capable of picking up a 43 kW load step with a 6 Hz frequency deviation. The main economic benefit of a large surge tank would be to reduce the running time of diesel engine/generators which are presently used to maintain frequency stability when picking up large pumps.

3 Note on Units

SI units are given for each variable in the nomenclature table. Computed values are in SI units. The input quantities were a mix of US and SI units. In equations with mixed units it is to be understood that US units are converted to SI units before the numeric calculations.

4 Surge Tank

In computing transients for designs with a surge tank we assumed that the changes in water level within the surge tank were small compared to the changes in pressure at the turbine. In selecting the surge tank size, economic factors may suggest a surge tank whose performance lies somewhere between the completely pressurized pipeline and the ideal surge tank considered here.

5 Pipe-Line Overview

The penstock was taken to be 318 ft long which included about 10 ft of conduit of unknown size between the new penstock and the turbine center line. The elevation of the water surface in the surge tank was taken to be about 145 ft and the turbine center line elevation 27 ft yielding a gross head of about 118 ft. Because air is presently admitted into the draft tube to avoid vibration the head recovery in the draft tube is unknown and was taken to be zero.

For the design without surge tank, we assumed the same gross head and a 1331 ft penstock which was either

1. all 36 in diameter, or
2. 318 ft of 36 in diameter and 1013 ft of 42 in diameter.

6 Turbine Properties

The large turbine ratings taken from the nameplate[4] are (alternate units in parenthesis)

1. Net head $H = 35.4\,\mathrm{m}\,(116\,\mathrm{ft})$
2. Power $P = 737\,\mathrm{hp}\,(550\,\mathrm{kW})$
3. Discharge $Q = 1.72\,\mathrm{m}^3/\,\mathrm{s}\,\left(60.7\,\mathrm{ft}^3/\,\mathrm{s}\right)$
4. Rotational speed $N = 900$ rpm

The turbine specific speed (rpm, hp, ft) is

$$N_s = N\frac{\sqrt{P}}{H^{5/4}} = 900 \text{ rpm}\frac{\sqrt{737\,\mathrm{hp}}}{(116\,\mathrm{ft})^{5/4}} = 64 \text{ rpm}.$$

This is a "medium" speed Francis turbine whose discharge is primarily a function of gate position and pressure independent of the rotational speed. If we assume further that the discharge coefficient of the turbine decreases linearly with time for an emergency shutdown, we can use Allievi's charts to estimate the magnitudes of the pressure transients.[5]

7 Pipe Properties

The designs under consideration include 36 in and 42 in steel pipe with 1/4 in wall thickness. Table 2 lists the basic pipe parameters. The first 3 columns show the outer diameter D, wall thickness e, and the modulus of elasticity E under tension in the hoop direction. The last three columns show the computed volume modulus K_V, effective bulk modulus K_E, and pressure wave velocity a.

D in	e in	E G Pa	K_V G Pa	K_E G Pa	a m/ s
36	0.25	210	1. 47	0.871	933
42	0.25	210	1.26	0.793	891

Table 2: Pipe parameters.

The pipe volume modulus K_V was computed from

$$K_V = \frac{Ee}{(D-e)}$$

where E is the elastic modulus in the hoop direction, e is the wall thickness, and $(D-e)$ is the diameter of the pipe at the middle of the pipe wall. The effective bulk modulus K_E was calculated from

$$K_E = \left(\frac{1}{K_W} + \frac{1}{K_V}\right)^{-1}$$

where $K_W = 2.14\mathrm{G\,Pa}$ is the bulk modulus of water at $10\,^\circ\mathrm{C}$ ([6] page 15) and K_V is the volume modulus of the pipe. The pressure wave propagation velocity a was computed using the formula

$$a = \sqrt{\frac{K_E}{\rho}}$$

where the $\rho = 1000\,\mathrm{kg/\,m^3}$ is the density of water at $10\,^\circ\mathrm{C}$.

8 Surge Tank and 36 in Steel Penstock

For initial discharge of $Q_0 = 80\,\mathrm{ft^3/\,s}$ the water velocity is

$$V_0 = \frac{Q_0}{A} = \frac{Q_0}{\pi/4 \times (D-2e)^2} = \frac{80\,\mathrm{ft^3/\,s}}{\pi/4 \times (35.5\,\mathrm{in})^2} = 3.\,55\,\mathrm{m/\,s}$$

The pipe-line constant is

$$\sigma = \frac{aV_0}{2gH_0} = \frac{933\,\mathrm{m/\,s} \times 3.\,55\,\mathrm{m/\,s}}{2 \times 9.81\,\mathrm{N/\,kg} \times 118\,\mathrm{ft}} = 4.7.$$

From the Allievi chart[5], $\theta \approx 24$ for a 20% pressure rise and the gate travel time is

$$T_G = \frac{2L\theta}{a} = \frac{2 \times 308\,\mathrm{ft} \times 24}{933\,\mathrm{m/\,s}} = 4.8\,\mathrm{s}.$$

9 No Surge Tank and All 36 in Steel Pipe

With no surge tank and 1331 ft uniform 36 in penstock, the calculations are the same as above except for the gate travel time which will now be

$$T_G = \frac{2L\theta}{a} = \frac{2 \times 1331\,\mathrm{ft} \times 24}{933\,\mathrm{m/\,s}} = 21\,\mathrm{s}.$$

10 No Surge Tank 36 in and 42 in Steel Pipe

To compute the pipe-line constant for a series connection of two different pipe diameters we note that the pipe-line constant can be written as

$$\sigma = \frac{aV}{2gH} = \frac{1}{2L/a}\frac{LV}{gH} = \frac{T_W}{T_2}$$

where

$$T_2 = 2L/a$$

is the time for a pressure wave to travel twice the length of the pipe and

$$T_W = \frac{LV}{gH}$$

is the hydraulic inertia constant or *water starting time*. For two series connected pipes with different pressure wave velocities a, lengths L, and water velocities V, the corresponding parameters are

$$T_2 = 2\left(L_1/a_1 + L_2/a_2\right)$$

and

$$\begin{aligned} T_W &= \frac{(L_1V_1 + L_2V_2)}{gH} \\ &= \frac{Q}{gH}\left(\frac{L_1}{A_1} + \frac{L_2}{A_2}\right) \end{aligned}$$

where Q is the plant discharge and A_1 and A_2 are the pipe cross sectional areas. For the case of 318 ft of 36 in diameter and 1013 ft of 42 in diameter pipes conveying $80\,\mathrm{ft^3/s}$ of water

$$T_2 = 2\left(\frac{318\,\mathrm{ft}}{933\,\mathrm{m/s}} + \frac{1013\,\mathrm{ft}}{891\,\mathrm{m/s}}\right) = 0.90\,\mathrm{s},$$

$$T_W = \frac{80\,\mathrm{ft^3/s}}{9.81\,\mathrm{N/kg} \times 118\,\mathrm{ft}}\left(\frac{318\,\mathrm{ft}}{\pi/4\,(35.5\,\mathrm{in})^2} + \frac{1013\,\mathrm{ft}}{\pi/4\,(41.5\,\mathrm{in})^2}\right) = 3.2\,\mathrm{s},$$

and

$$\sigma = \frac{T_W}{T_2} = \frac{3.2\,\mathrm{s}}{0.90\,\mathrm{s}} = 3.6.$$

From the Allievi diagram, $\theta \approx 22$ for a 20% pressure rise and the gate travel time is

$$T_G = T_2\theta = 0.900\,\mathrm{s} \times 22 = 20\,\mathrm{s}.$$

11 Governing Capability

Eliminating the surge tank not only increases the overspeed time on load rejection (from longer gate closing times) but also reduces the capability of the governor to control the turbine speed and hence line frequency. Here we quantify the changes in governing to be expected.

An approximate worst case for frequency control is the large unit operating alone near full power. The quality of speed control is determined by the relationship between the machine starting time T_M and water starting time T_W.The machine starting time is

$$T_M = \frac{\omega^2 I}{P} = \frac{(2\pi \times 900/60\,\mathrm{s})^2 \times 7000\,\mathrm{lb\,ft}^2}{550\,\mathrm{kW}} = 4.8\,\mathrm{s}$$

where ω is the rotational speed in radians/sec, I is the rotational moment of inertia for the generator rotor and flywheel[2], and P is the rated power. The water starting time with the new 36 in penstock and penstock at the rated flow for the large unit is

$$T_W = \frac{L}{gH}\frac{Q}{A} = \frac{330\,\mathrm{ft}}{9.81\,\mathrm{N/\,kg} \times 116\,\mathrm{ft}} \frac{1.72\,\mathrm{m}^3/\,\mathrm{s}}{\pi/4 \times (35.5\,\mathrm{in})^2} = 0.78\,\mathrm{s}$$

whereas without the surge tank and 42 in pipe replacing the flume, the water starting time would be

$$T_W = \frac{1.72\,\mathrm{m}^3/\,\mathrm{s}}{9.81\,\mathrm{N/\,kg} \times 119\,\mathrm{ft}} \left(\frac{318\,\mathrm{ft}}{\pi/4\,(35.5\,\mathrm{in})^2} + \frac{1013\,\mathrm{ft}}{\pi/4\,(41.5\,\mathrm{in})^2} \right) = 2.4\,\mathrm{s}$$

Note that the water starting time for 36 in pipe replacing the flume would be even larger.

The per unit speed change n following a per unit power change p will be approximately[7]

$$n = 2.5 \frac{T_W}{T_M} p.$$

Thus the Hz frequency deviation per kW load change with the surge tank would be

$$2.5 \times \frac{0.78\,\mathrm{s}}{4.8\,\mathrm{s}} \times \frac{60\,\mathrm{Hz}}{550\,\mathrm{kW}} = 0.044 \frac{\mathrm{Hz}}{\mathrm{kW}}.$$

A realistic maximum load step causing a 6 Hz deviation would be about 136 kW.

With the 42 in pipe replacing the flume, the Hz frequency deviation per kW load change would be

$$2.5 \times \frac{2.4\,\mathrm{s}}{4.8\,\mathrm{s}} \times \frac{60\,\mathrm{Hz}}{550\,\mathrm{kW}} = 0.14\frac{\mathrm{Hz}}{\mathrm{kW}}$$

and a realistic maximum load step causing a 6 Hz deviation would be about 43 kW.

References

[1] Memo from Bob Butera, HDR, regarding condition assessment and potential rehabilitation work required, June 10, 2005.

[2] *The Pelican, Alaska Hydroelectric Station*, Gene Newman, Waterpower '85, American Society of Civil Engineers, New York, NY, 1986.

[3] *Pelican Hydroelectric Facility Condition Assessment Field Report*, Polarconsult Alaska, Inc., June 30, 2006.

[4] Dan Hertridge, Polarconsult Alaska, Inc., personal communication, July 2006.

[5] *Water Hammer*, George R. Rich, Section 27 of *Handbook of Applied Hydraulics Third Edition,* edited by Calvin David and Kenneth Sorensen, McGraw-Hill, New York, 1969.

[6] *Analysis of Water Surge*, John Pickford,Gordon and Breach Science Publishers, New York, 1969.

[7] *Waterhammer and Governor Analysis*, Larry Clifton, Water Power & Dam Construction, August 1987.